Investigation on Noise of Rotary Compressors using Fluid-Structure Interaction

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ABSTRACT

The noise generated in the rotary compressor can be classified by pressure pulsation of the refrigerant and structural vibration. This paper deals with the noise generated by pressure pulsation of refrigerant in a rotary compressor. During the operation of the compressor, the refrigerant and the internal structure of the compressor have strong interaction with each other. At this time, the oil around the reed valve causes the compressor to overpressure the refrigerant, which is the main cause of noise generation. Therefore, the interaction between the refrigerant and the valve during the compressor operation is analyzed using the FSI technique. The pressure pulsation of the refrigerant and the behavior of the valve were analyzed. Also, the noise characteristics were confirmed through the spectrum analysis of the generated noise.

1. INTRODUCTION

The air conditioner used in the home has been developed and improved by core technologies such as compressor, heat exchanger, high performance of fan, and cycle optimization control technology. In order to cope with the changing trend of this environment, noise and vibration reduction technology is one of the key technologies.

As for the noise reduction of the compressor up to now, the rotary compressor which is most used in the air conditioner, Sano and Mitsui (1984) studied the pressure pulsation generated when the refrigerant is compressed in the internal space of the compressor and noise caused by the resonance frequency of the internal space, noise due to collision or sliding between parts. Johnson and Hamilton (1972) specifically discusses the characteristics of the noise generated between the spatial resonance sound and the refrigerant gas discharged from the discharge valve.

Among the various methods of reducing noise in a rotary compressor, a method using a resonator and a muffler is mainly used. Many researchers have published a number of results on the results of a drastic reduction of noise by installing a small resonator near the discharge port. As for the muffler, the research has been conducted on the characteristics of the dipole type muffler by reducing the spatial resonance sound. In recent years, computation analysis as well as experimentation have also been actively made for noise analysis of a rotary compressor and improvement of performance.

With these results, many of the results that have been interpreted and reviewed so far have been applied to products. However, there is a tendency to be in agreement with the results under actual use conditions. This interpretation needs to be considered in terms of characteristics that are accompanied by pressure pulsations as turbulence. In addition, various methods are used to reduce noise, but qualitative analysis through experiments is mainly due to complex physical phenomena.

In this paper, the interaction between the refrigerant and the internal structure of the rotary compressor is analyzed by using the FSI method considering the fluid-structure interaction. In addition, the noise source characteristics of each component were analyzed through comparison of various driving speeds.

2. NUMERICAL MODEL

2.1 Twin Rotary Compressor

In this study, the twin rotary compressor was used as a model (Figure 1). The twin rotary compressor has two compressing cylinders, each having a phase difference of 180 degrees, which is advantageous in that the torque is smaller than that of a single rotation compressor. The compressed refrigerant in the two compression cylinders pushes out the valve attached to each compression cylinder and is discharged through the muffler on one side. The

refrigerant then passes through an upper chamber with a stator/rotor and is discharged through a pipe attached to the top of the compressor.
The twin rotary compressor has two mufflers, which have a shape with several rooms because of compressor structural reasons. The upper muffler and the lower muffler in this model have a shape divided into five and three chambers, respectively, and can affect the flow of the discharged refrigerant.

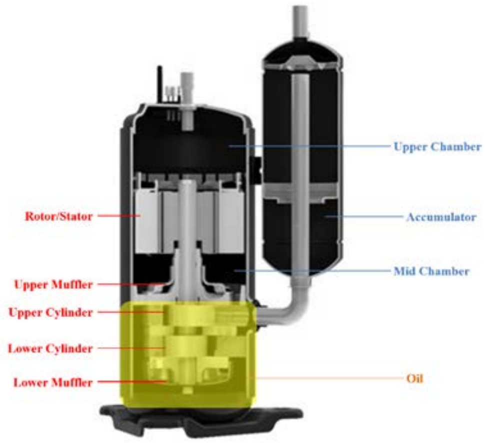


Figure 1: Twin rotary compressor

The rotary compressor is operated under ARI conditions. Also, as one of the methods for improving the efficiency of the domestic air conditioner, development of a product using an inverter system in which the operation speed of the compressor is changed according to driving conditions is actively being developed. It is important to understand the behavioral characteristics of the main factors because the flow characteristics may vary depending on various operating conditions when the inverter system is applied. Therefore, to analyze the characteristics according to driving conditions, two driving speeds are analyzed (Table 1).

Table 1: Main Specification of the compressor

Type of compressor	Twin rotary compressor
Refrigerant	R410a
Suction pressure	9.12 kg / cm^2
Discharge pressure	33.45 kg / cm^2
Driving condition	40, 60 rps

2.2 Numerical Methods

In the rotary compressor, the refrigerant is compressed in an isolated cylinder covered with a valve, and the compressed high-pressure refrigerant is discharged by pushing the valve. the moving grid is used because the computational grid of the fluid region is changed according to the displacement of the discharge valve. Fluid-Structure Interaction(FSI) provides the exact solution to common natural phenomena by exchanging the respective results based on two independent solvers. In this study, governing equations are RANS(Reynolds-Averaged Navier-Stokes) equations. As the turbulence model of the flow analysis, k-w sst model which have high accuracy in the boundary layer is used.
When the refrigerant is discharged from the cylinder, over compression occurs due to the oil stiction force inside the compressor. A typical compressor valve model is shown in Figure 2. The over compression of the refrigerant must be considered because it has an influence on the pressure pulsation. The factors affecting the opening time of the valve are viscosity of oil, height of oil, size of valve plate and valve seat, spring constant. We used the model of Bukac (2002) to consider the above factors in order to predict the exact discharge time.

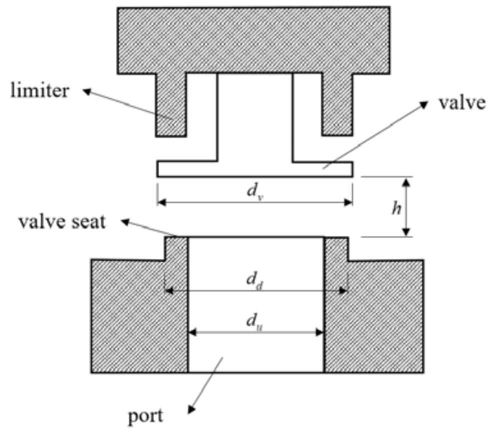


Figure 2: Generic model of a compressor valve

$$F_{adh} = C \cdot \frac{\dot{h}_{oil}}{h_{oil}{}^{3}} \tag{1}$$

$$C = \frac{3\pi\mu\left(r_B{}^4 - r_A{}^4\right)}{32}\left(\frac{r_B{}^2 - r_A{}^2}{\left(r_B{}^2 + r_A{}^2\right)\left(\ln r_B - \ln r_A\right)} - 1\right) \tag{2}$$

$$r_A = \frac{d_u}{4}\left(1 + \frac{h_0}{h}\right) + \frac{d_d}{4}\left(1 - \frac{h_0}{h}\right) \tag{3}$$

$$r_B = \frac{d_u}{4}\left(1 - \frac{h_0}{h}\right) + \frac{d_d}{4}\left(1 + \frac{h_0}{h}\right) \tag{4}$$

3. RESULT

Numerical analysis is performed using an unstructured mesh. The number of grid in the flow region was about 1 million, and the grid was concentrated in the region of the compression cylinder and the muffler. The number of grid of the solid region (valve part) was about 20,000. In order to accurately simulate the pressure pulsation, the time step size was selected as the time when one cycle was divided into about 840 times (40rps case's time step size = 3.0e-05s, 60rps case's time step size = 2.0e-05s).
The pressure of the compression cylinder during the compressor operation was analyzed. As shown in the Figure 3, the refrigerant begins to compress at the design suction pressure and the pressure rises as the crankshaft rotates. It can be seen that the refrigerant is not discharged directly from the design discharge pressure but is discharged after some over compression and is converged to the design discharge pressure. The overpressure is 199,000 pascal for 40rps and 225,000 pascal for 60rps. Ideally, the valve is opened when the internal pressure of the compression cylinder is larger than the discharge pressure, but the discharge is delayed by the oil stiction.

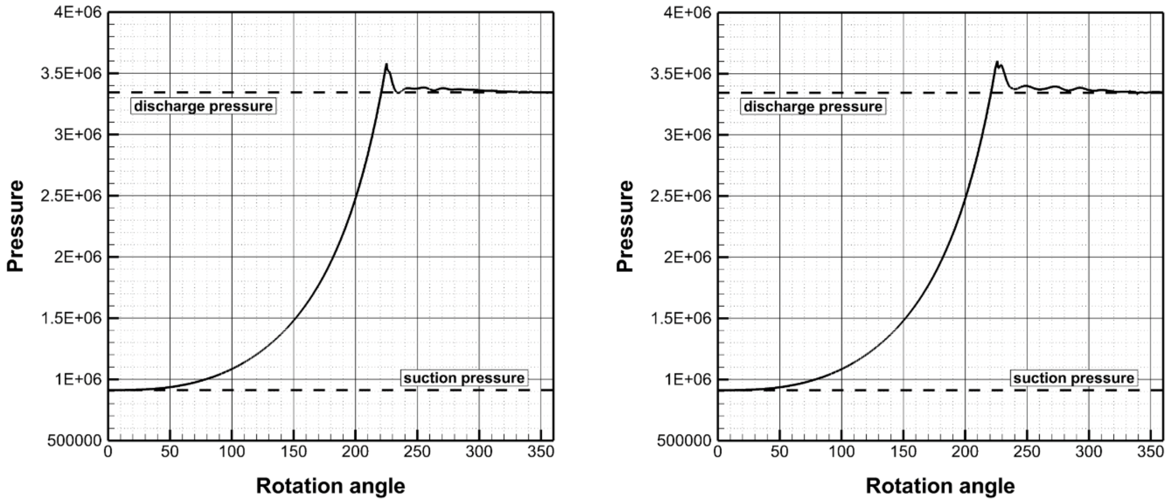


Figure 3: Pressure at compression cylinder (left: 40rps, right: 60rps)

The behavior of the valve according to the rotating angle of the rotary compressor has been simulated. Figure 4 shows the displacement of the valves according to the driving speed. The upper valve and the lower valve have a phase difference of 180 degrees because of the phase difference of the compression cylinder. The displacement of valve has two peaks in common. The first peak of the valve occurs due to the over-compressed refrigerant discharged and moves to maximum causing the valve to hit the retainer. The second peak occurs when the refrigerant discharged by the subsequent compression process push the valve again. As for the driving speed, the faster the driving speed, the larger the displacement is on average.

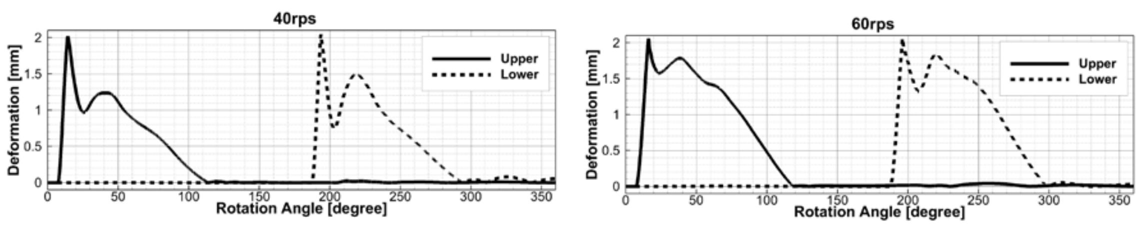


Figure 4: Valve Deformation (left: 40rps, right: 60rps)

To analyze the noise generated by the compressor, the pressure at the bottom of the rotor was analyzed. Figure 5 shows the pressure from the start of discharge in the upper compression cylinder to the completion of discharge of the lower compression cylinder. As can be seen in the figure, it shows a strong pressure fluctuation when refrigerant is discharged from the upper compression cylinder and a weak pressure fluctuation when refrigerant is discharged from the lower compression cylinder. This is because the refrigerant discharged from the upper cylinder is discharged through the upper muffler while the refrigerant discharged from the lower cylinder is discharged through the lower muffler and upper muffler, likes two stage muffler.

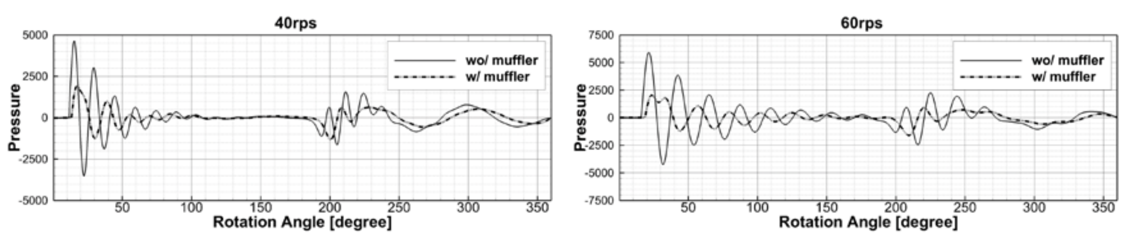
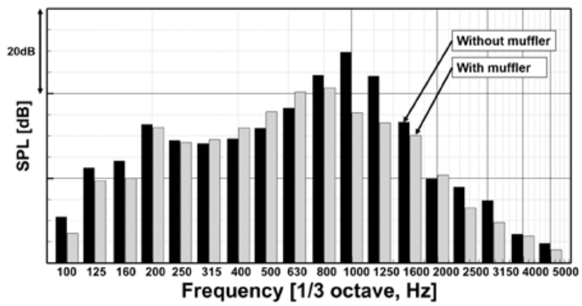


Figure 4: Pressure at bottom of the rotor (left: 40rps, right: 60rps)

The results of the FFT analysis on the pulsation analysis in the rotor lower space show that resonance components are commonly concentrated around 200 Hz and 800 to 1000 Hz (Figure 5). It can be seen that this is the resonance mode(200Hz) at the space divided into the upper and lower parts of the rotor in the internal structure of the compressor, and the resonance mode(1000Hz) by the compression cylinder discharge part. When the muffler is added, the resonance mode by the discharge part of the compression cylinder is changed, and it can be confirmed that the component near 1000 Hz is reduced.



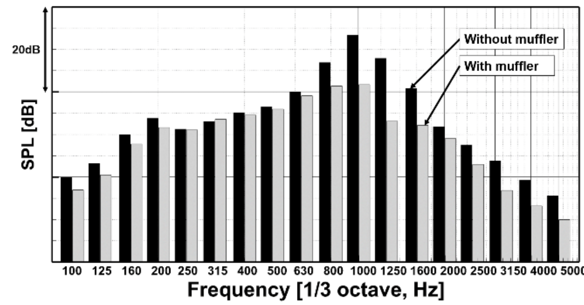


Figure 5: Sound Pressure Level at bottom of the rotor (left: 40rps, right: 60rps)

4. Conclusions

In this paper, Fluid-Structure Interaction analysis was conducted to understand the noise generation mechanism of rotary compressor used in air conditioner. The analytical results of the main factors such as the noise inside the compressor, the cylinder pressure and the displacement of the discharge valve were verified by comparing with the experimental results. In order to understand the noise characteristics of the interpreter compressor, the characteristics of the main factors in the compressor were analyzed varying the operating speed.

As a result of examining the main factors influencing the noise of the compressor, a noise is generated by the internal pressure wave when the discharge valve is opened, and a vortex is generated according to the behavior of the discharge valve. As a result of the FFT analysis, 200 Hz and 800 to 1000 Hz components are generated in the compressor, and these correspond to the acoustic resonance and discharge valve modes, respectively. As the operation speed increases, the overpressure axis also becomes higher and the behavior of the discharge valve changes greatly.

NOMENCLATURE

C	stiction coefficient
F_{adh}	adhesion force
d_d	outer seat diameters
d_u	inner seat diameters
h_0	initial thickness of the oil
h_{oil}	thickness of the oil

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